ISSN: 2449 – 0539
BAYERO JOURNAL OF ENGINEERING AND TECHNOLOGY (BJET) VOL.14 NO.2, pp 169-182, AUGUST, 2019

MODIFICATION OF LOCALLY DEVELOPED GROUNDNUT SHELLER

Muhammad, A. I [1], and Isiaka, M [2]

[1] Department of Agricultural and Environmental Engineering, Bayero University Kano, P.M. B. 3011, Kano, Nigeria.
[2] Department of Agricultural Engineering, Ahmadu Bello University, Zaria, Nigeria.

Corresponding author's email address: aimuhammad.age@buk.edu.ng

ABSTRACT

A locally constructed groundnut sheller (Dan yamel) was modified. The modified sheller consist of shelling unit which comprised of a hopper with feed control and shelling chamber made of a rasp bar, a cleaning unit, and a prime mover. The sheller operates on a single belt drive system. The performance test of the modified sheller was carried out using Ex-Dakar groundnut variety with a moisture content of 8%. A feed rate of 300 kg/h and shelling speed of 180 rpm were used during the test. The performance indices obtained were significantly improved after the modification except for the output capacity, which was slightly increased. The modified sheller has an output capacity, shelling efficiency, cleaning efficiency, mechanical damage, and scatter loss of 239.81 kg/h, 98.32%, 50.63%, 4.33 %, and 3.24%, respectively as against 233.18 kg/h, 86.19%, 8.11%, and 9.52%, respectively recorded with the old sheller.

KEYWORDS: Groundnut sheller; groundnut decorticator; shaft design; blower design; performance evaluation

1.0 INTRODUCTION

Groundnut (*Arachis hypogaea*) is grown for its edible oil and protein-rich seeds as an annual crop in tropical and subtropical regions of the world. Groundnut belongs to the family *Leguminosae* (Asiedu, 1992). It is the sixth most crucial oil-producing crop in the world with production concentrated in Asia and Africa recording 56 and 40 % of the global area and 68 and 25 % of the global production, respectively (Ntare, Diallo, Ndjeunga, and Waliyar, 2008). India, China, USA, Brazil, Senegal, and Nigeria were the global producers of about 80% of the crop. Nigeria recorded about 907 kg/ha of groundnut kernels yield annually (Ibrahim and Onwualu, 2005). Groundnut production in Nigeria nowadays is at the subsistence level. The traditional shelling of groundnut is carried out manually which involves a lot of drudgery, time-consuming, unhygienic conditions, and low output (Muhammad et al., 2015a). The traditional shelling involves the breaking down of groundnut pods by applying pressure on the pods using fingers as a result of which the kernels are exposed. This method is inefficient, and the output depends on individual health and motivation and ranges from 1 – 2 kg/h to 10 – 15 kg/h (Ramli, 2003). Although this method is not efficient but yielded an excellent result in terms of kernel damages, other methods are beating the heap of groundnut on the hard floor with sticks or use of pestle and mortar. Both methods are associated with kernel damage and unhygienic condition of kernel due to the presence of impurities such as sand (IAR, 2005). Several groundnut shelling equipment were reported in literature ranging from hand-operated decorticators to sophisticated machines. Ikechukwu and co-researchers developed groundnut sheller that comprised of the hopper, shelling chamber, cleaning chamber and a blower unit. The performance parameters reported were output capacity of 400 kg/h, shelling efficiency of 95.25%, cleaning efficiency of 91.67% and mechanical damage of 17.25% (Ugwuoke, Okegbile, and Ikechukwu, 2014). Gitau, Mboye, Njoroge, and Mburu (2013) modified and tested two manually operated groundnut decorticators (Wooden Beater Sheller and Rod Beater Sheller).

In Nigeria, numerous groundnut shellers are fabricated by roadside fabricators. From our survey (data not shown), these shellers range in capacities from 25 – 120 kg/h with high kernel damage sold at exorbitant prices. Among these shellers, '*Dan yamel*' (a popularly used groundnut sheller in Dawanau market) was chosen for modification based on its performance indices (shelling efficiency of 89.23%, kernel breakage and scatter loss of 17.81% and 13.73%, respectively). Besides, the machine has no cleaning unit. In view of the aforementioned, this study aimed to modify and improve the performance of the existing sheller and to incorporate a cleaning unit in the modified groundnut sheller to maximize its performance.

2.0 MATERIALS AND METHODS

2.1 Design Consideration

Establishment of design consideration is vital for the optimum performance of groundnut sheller. The following design considerations were adopted:

i. Engineering properties of groundnut pods and kernels were taken into consideration.
ii. The use of locally available materials and technology for construction.
iii. Low cost of construction.
iv. Petrol engine was chosen as the prime mover due to epileptic power supply.
v. Drum and blower speeds of 300 and 680 rpm, respectively were adopted from the old sheller for pulley design.
vi. A rasp bar shelling drum of 250 mm diameter (D_2), 450 mm length 500 mm shelling chamber length were adopted from the old sheller.

2.2 Description of the modified Sheller

The modified sheller consisted of the shelling unit, cleaning unit, delivery, and discharge chutes. The shelling unit comprised of a rasp bar made from an iron rod as against the rasp bar from angle iron. This was changed in order to reduce the severe kernel damages in the old machine. Also, the concave was made from a cylindrical iron rod. The cleaning unit consisted of a centrifugal blower constructed from the metal iron sheet. The delivery chute (seed discharge) and discharge chute (chaff outlet) were constructed from metal sheet. The delivery chute was slightly tapered at an angle of 28^{o} toward the machine base to ensure smooth kernel discharge by gravity. The isometric and orthographic view of the sheller is exhibited in Figure 1a and b.

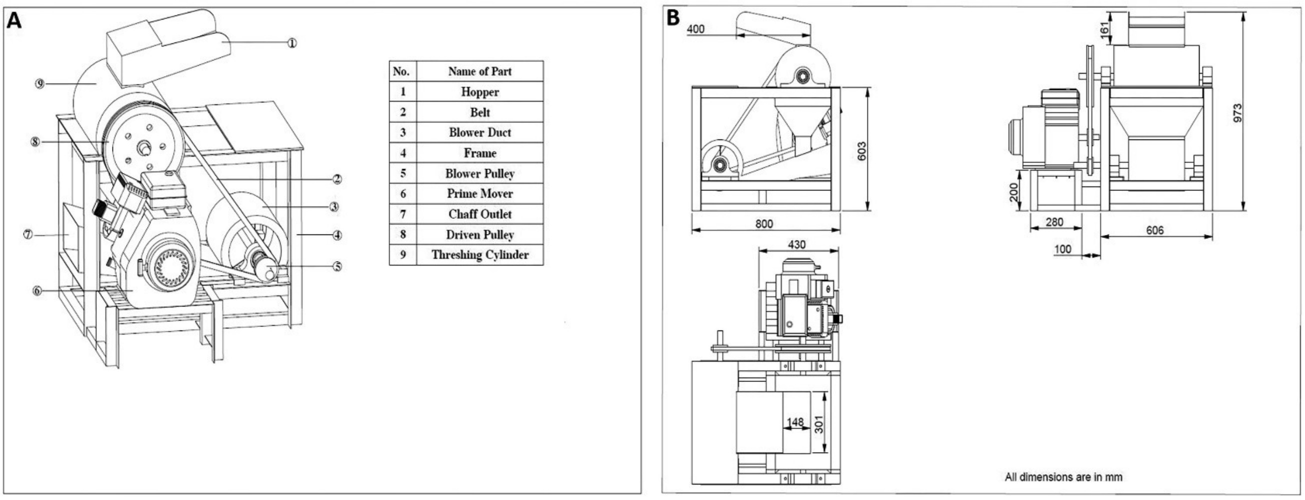

Figure 1: Isometric (A) and orthographic view (B) of the modified groundnut sheller

2.3 Determination of size of Machine components

2.3.1 Determination of size of the hopper

The hopper was redesigned to accommodate more groundnuts using the relation below:

$V_m = L \times W \times h$ … (1)

Where, L = Length, W = Width and h = height, L = 90% of drum length = 0.9 × 450 mm = 405 mm, W = 500 mm, h = 120 mm

Therefore, V_m = 405 × 500 × 120 = 2.43 × 107 mm^3

The angle of inclination from horizontal is ≥ 26^o as determined by Odesanya, Adebiyi, and Salau (2015).

2.3.2 Determination of size of the concave

The concave radius was determined following Nalado (2006):

$R_C = R_D + H_P + C_C$ … (2)

Where, R_C = Concave radius (mm), R_D = Radius of drum; Recall D_2 = 250 mm

$R_D = \frac{D_2}{2} = \frac{250}{2}$ = 125 mm

H_P= 14 mm; Height of rasp bar (14 mm rod was used), C_C = 14 mm; concave clearance was obtained based on the geometric mean diameter of groundnut kernels from our previous study (Muhammad et al., 2015b).

$\therefore R_C = 125 + 14 + 14$ = 153 mm

Therefore, the concave diameter is 153 × 2 = 306 mm, length of concave = 500 mm was obtained in the design consideration, diameter of concave openings = 12 mm based on the geometric mean diameter of groundnut kernels from our previous study, and concave clearance = 14 mm (Muhammad et al., 2015b).

2.3.3 Determination of the volume of shelling chamber

The capacity of the shelling chamber was computed based on its cylindrical geometry as:

$V_{SC} = \frac{\pi \times D_{SC}^2 \times L_{SC}}{4}$ … (3)

Where, V_{SC} = volume of shelling chamber (mm^3), L_{SC} = Length of shelling chamber (500 mm)

D_{SC} = Diameter of shelling chamber = D_2 + 2 C_C = 250 + 2 x 14 = 278 mm

$V_{SC} = \frac{\pi \times 278 \times 500}{4}$ = 109170.345 mm^3

2.3.4 Design for Blower components

The design for various components of the blower is as follows:

a. *Determination of air flow rate*

Air discharge by a centrifugal blower (Q) can be easily estimated based on air velocity required for cleaning, depth of air stream through the duct, and width of the duct. Q is obtained from the relation given by Joshi (1981) in equation 4:

$$Q = AV \qquad \ldots \qquad (4)$$

Where, Q = air flow rate (m^3/s), V = velocity of air required for cleaning obtained from literature (m/s) = 9.8 m/s (Ghanem and Shetawy, 2009), A = Area of air duct (m^2), $A = L \times B = 0.46 \times 0.135$ m^2.

$\therefore Q = 0.46 \times 0.135 \times 9.8 = 0.6086$ m^3/s

b. Determination of specific speed, N_s of blower

The specific speed of the blower was calculated using the relationship from (Oyelami, Olaniyan, Iliya, and Idowu, 2008) as:

$$N_s = \frac{N_1\sqrt{Q}}{H^{3/4}} \qquad \ldots \qquad (5)$$

Where, Q =Air discharge in gallons per min (gpm) = 0.6086 m^3/s × $60\,\frac{m^3}{min} \times 1127 g/m^3$ = 41153.566 gpm, N_1 = motor speed (1500 rpm), g = acceleration due to gravity (9.81m/s^2), H = Total head imparted to the flow by the blower, $H = \frac{V^2}{2g} = \frac{9.8^2}{2\times 9.81}$ = 4.895 m = 4.895×3.281 = 16.0597 ft.

$$\therefore N_s = \frac{1500 \times \sqrt{41153.566}}{16.0597^{3/4}} = 37930 \approx 38000\ /min$$

The pressure (ψ) and flow coefficients ($\emptyset$) were obtained using the specific speed, N_s = 38000/min for straight radial centrifugal blower from (Sahay and Singh, 2007) as ψ = 0.9 and $\emptyset$ = 0.51.

c. Determination of blower outlet diameter

The outlet (D_o) and inlet (D_i) diameters of the blower were computed according to equation 6 and 7 (Gui and Xi, 2010):

$$D_o = \sqrt[3]{\frac{240Q}{\pi^2 \psi N_1}} \qquad \ldots \qquad (6)$$

Recall that N_1= 1500 rpm ψ = 0.9, Q = 0.6086 m^3/s

$$\therefore D_o = \sqrt[3]{\frac{240 \times 0.6086}{\pi^2 \times 0.9 \times 1500}} = 0.2221 \text{ m} \approx 222 \text{ mm}$$

$$D_i = 1.2 \times \emptyset^{1/3} \times D_o \qquad \ldots \qquad (7)$$

Recall, $\emptyset$ = 0.51, $\therefore D_i = 1.2 \times 0.51^{1/3} \times 0.2221$ = 0.2129 m ≈ 213 mm

d. Determination of the number of blades

The number of blades (N_b) was computed as reported in (Muhammad, 2009) and (Joshi, 1981):

$$N_b = \frac{q}{v} \qquad \ldots \qquad (8)$$

Where, v = volume of air displace per sec., q = flow rate (m^3/s)

$$q = AV \qquad \ldots \qquad (9)$$

A = area of duct inlet for the aspirator = $W \times D_e$, and W = width over which air is required = 0.135 m (obtained from duct width), D_e = equivalent diameter of airflow rate (m).

$$D_e = 1.265\sqrt{\frac{(ab)^2}{a+b}} \qquad \ldots \qquad (10)$$

Where, a and b are length and width of air plenum, respectively; a = 0.55 m, b = 0.25 m

$$D_e = 1.265\sqrt{\frac{(0.55 \times 0.25)^2}{0.55+0.25}} = 0.1945 \text{ m}$$

$\therefore A = W \times D_e = 0.135 \times 0.1945 = 0.0895$ m^2

According to Sahay and Singh (2007), the velocity of air (V) required for cleaning ≤ terminal velocity of groundnut kernels, and it ranged between 9.4 - 17.8 m/s. Also, Ghanem and Shetawy (2009) reported that an air stream of 9.8 - 9.9 m/s was sufficient to separate 98.5 and 96.5 % of shells, fine particles, and dust from the kernels. As such 9.8 m/s was selected for V.

Now substituting air velocity of 9.8 m/s in equation (9) above we have

$q = 0.0895 \times 9.8 = 0.8767$ m^3/s

In order to get the actual flow rate (q_a), 30% efficiency was assumed (Muhammad, 2009)

$$q_a = \frac{q}{0.3} = 2.9222 \text{ m}^3/\text{s} \quad \ldots \quad (11)$$

The volume of air displaced by the blade, V^1 was computed, according to Muhammad (2009) and Joshi (1981) as:

$$V^1 = \frac{\pi\, W\, D_e}{4} \quad \ldots \quad (12)$$

$$V^1 = \frac{\pi \times 0.46 \times 0.1945}{4} = 0.0703 \text{ m}^3/\text{blade}$$

The relationship for blade revolution per sec was adopted from Muhammad (2009) and Joshi (1981):

$$\text{Blade (rev/sec)} = \frac{V}{\pi\, D_e} \quad \ldots \quad (13)$$

$$\text{Blade (rev/sec)} = \frac{9.8}{\pi \times 0.1945} = 16.0382 \text{ rev/sec}$$

$$v = \text{Blade (rev/sec)} \times V^1 \quad \ldots \quad (14)$$

Where, v = volume of air displace per sec,
$v = 16.0382 \times 0.0703 = 1.1275$ m^3 rev/sec blade

Recall that, number of blades,

$$N_b = \frac{q}{v} = \frac{2.9222}{1.1275} = 2.5918 \approx 3 \text{ blades}$$

e. Determination of blower width

Sahay and Singh (2007) proposed the width of the blower as:

$$W = \frac{175Q}{\emptyset\, N_1 D_o^2} \quad \ldots \quad (15)$$

Where, W = width of the blower (inch), recall Q (cfm) = 0.6086 m^3/s = 1289.552 cfm, D_o (in) = 0.2221 m = 8.7441 in, $\emptyset = 0.51$

$$\therefore W = \frac{175 \times 1289.552}{0.51 \times 1500 \times 8.7441^2} = 3.8582 \text{ in} = 0.098 \text{ m}$$

2.3.5 Determination of pulley sizes

Due to the incorporation of the cleaning system, the pulley sizes need to be redesigned. The sizes of the driven, driving and blower pulleys were determined according to equation 16 (Karwa, 2010):

$$\frac{D_{p2}}{D_{p1}} = \frac{N_1}{N_2} \quad \ldots \quad (16)$$

Where, D_{p1} = effective diameter of the driving pulley (75 mm), D_{p2} = effective diameter of the driven pulley, N_1 = motor speed (rpm), N_2 = shelling speed (300 rpm) and N_3= blower speed (680 rpm) as adapted from Butts, Sorensen, Nuti, Lamb, and Faircloth (2009).

$$N_1 = \frac{P \times 60}{2\pi T} \quad \text{(Khurmi \& Gupta, 2005)} \ldots (17)$$

P = 4.1 kW; T = 25.4973 Nm (obtained from prime mover operators' manual).

$$N_1 = \frac{4.1 \times 10^3 \times 60}{2 \times \pi \times 25.4973} = 1535.54 \approx 1536 \text{ rpm}$$

$$D_{p2} = \frac{1536 \times 75}{300} = 384 \text{ mm.}$$

The standard nearest pulley size selected was 375 mm (Karwa, 2010).

The corresponding pulley size for the blower is calculated as:

$$D_{p3} = \frac{N_1 \times D_{p2}}{N_3} = \frac{300 \times 375}{680} = 165.44 \text{ mm.}$$

The standard nearest pulley size selected was 160 mm (Karwa, 2010).

2.3.6 Determination of belt length

The length of belt required for the shelling drum and blower was computed from equation 18 (Mohammed and Hassan, 2012):

$$L = 2c + \pi\frac{(D_{p2} + D_{p1})}{2} + \frac{(D_{p2} + D_{p1})^2}{2c} \quad \ldots (18)$$

Where, L = belt length (mm), c = distance between the driving and the driven pulleys (700 mm).

$$L = 2 \times 700 + \pi\frac{(375+75)}{2} + \frac{(375+75)^2}{2 \times 700}$$
= 2251.501 mm ≈ 2252 mm

The nearest standard pitch length selected was 2286 mm (Khurmi and Gupta, 2005).

2.4 Determination of machine Power requirement

This is the summation of the estimated power requirement for the shelling unit, centrifugal blower, and pulley.

2.4.1 Determination of power required to shell groundnut pods

The power required to shell the groundnut pods was calculated as reported in Alonge and Kosemani (2011) as:

$P_r = T\omega$ ... (19)

Where, T = Toque (Nm) $= \frac{\pi D_2{}^3 t}{16}$, P_r = power required to shell groundnut pods (W), recall D_2 = 0.250 m, t = allowable shear stress of shelling shaft (56.25 x 10^{-3} N/m^2), ω = angular speed = $\frac{2\pi N_2}{60}$ = 31.4159, N_2 = shelling speed (300 rpm)

$$\therefore P_r = \frac{\pi \times 0.250^3 \times 56.25 \times 10^{-3} \times 31.4159}{16}$$

$= 5.422 \times 10^{-3}$ W

2.4.2 Determination of power to overcome rotation

The power to overcome air resistance against the rotation of the shelling drum and frictional force in the bearing was obtained from the relation given by Mohtasebi, Lar, Alidadi, and Besharati (2006) as:

$P_f = AV_L + BV_L$ … (20)

Where, P_f = power to overcome rotation (N), A = 0.85 – 0.90 N per 100 kg mass of rasp bar type sheller cylinder and 5 – 5.5 N per 100 kg mass of pegs type, B = (drum diameters of 550 mm) = 0.065 Ns^2/m^2 per m of drum length of rasp bar type and 0.045 Ns^2/m^2 for peg type, V_L = linear speed of belt in m/s determined as: $V_L = \frac{\pi D_2 N_2}{60}$ (Karwa, 2010).

Recall, D_2 = 375 mm, N_2 = 300 rpm, A = 1.7 N (20.387 kg drum), B = 0.065 Ns^2/m^2

$V_L = \frac{\pi \times 0.375 \times 300}{60}$ = 5.89 m/s

$\therefore P_f = 1.7 \times 5.89 + 0.065 \times 5.89$

= 660.3959 W

2.4.3 Determination of power required to drive the blower

The power required to drive the centrifugal blower was determined following Alonge and Kosemani (2011) as:

$P_b = W_B R\omega$ … (21)

Where, P_b = power required to drive the blower (W), W_B = weight of blower (35.3154 N), R = radius of blower (m) = 0.2129 ÷ 2

$$\therefore P_b = 35.3154 \times \frac{0.2129}{2} \times \frac{2\pi \times 680}{60}$$

= 267.6994 W

2.4.4 Determination of power required to drive the shelling drum

The power required to drive the shelling drum was exhibited in equation 22 (Alonge and Kosemani, 2011):

$P_p = W_C R_S \omega$ … (22)

Where, P_p = power required to drive the shelling drum (W), R_S = radius of the shelling drum (0.250 ÷ 2), ω = 31.4159, W_C = weight of drum rasp bars + weight of shelling shaft (N) = 207.189 + 199.996 = 407.185 N

$$\therefore P_p = 407.185 \times \frac{0.250}{2} \times 31.4159$$

= 1599.026 W

The total power, P_e required for the whole machine was:

$P_e = P_r + P_f + P_b + P_p$ … (23)

P_e = 5.422 x 10^{-3} W + 660.3959 +267.6994 W + 1599.026 W = 2527.127 W ≈ 2.527 kW

To determine the required capacity of the prime mover, the transmission efficiency was considered (Mohammed and Hassan, 2012):

$P_m = \frac{P_e}{\eta}$ … (24)

Where, P_m= power rating of the prime mover to be used (kW), P_e = summation of power required to drive the shelling unit and centrifugal blower (2.527 kW), η = assumed drive efficiency (75% efficiency was assumed).

$$P_m = \frac{2.527}{0.75} = 3.369 \text{ kW} \approx 3.4 \text{ kW}$$

The nearest prime mover available in the market was 4.1 kW, thus selected.

2.5 Design of Shaft

The stress present on the shaft could be in the form of bending, axial, and torsional stresses. Therefore, the shaft material used was from a ductile material (mild steel of grade C1030). The subsequent sections discussed the design of the shelling and blower shafts.

2.5.1 Shelling shaft

The shelling shaft was subjected to both bending and torsional loads. The bending load is due to weight and moments of the shelling shaft. The torsional moment was determined following Shigley and Mischke (2001).

$$M_t = 9.55\frac{H}{N_2} \quad \dots \quad (25)$$

Where, M_t = torsional moment (N), H = rated power of engine (4.1 kW), N_2 = shelling drum speed (300 rpm).

$$M_t = 9.55\frac{4.1 \times 10^3}{300} = 130.517 \text{ Nm}$$

Also, the tension on the belt was calculated from Karwa (2010).

$$M_t = (T_1 - T_2) \times r \quad \dots \quad (26)$$

$$\therefore M_t = (T_1 - T_2) \times r = 130.517 \text{ N}$$

Where, T_1 = tension on tight side (N), T_2 = tension on slack side (N), r = radius of driven pulley = 375 ÷ 2 = 187.5 mm.

The angle of contact made by the belt on the pulley was obtained from Khurmi and Gupta (2005).

$$2.3log\left(\frac{T_1}{T_2}\right) = \mu \times \theta \times Cosec\beta \quad \dots \quad (27)$$

Where, μ = frictional coefficient of belt and pulley (0.4), θ = groove angle, β= angle of wrap $= \frac{1}{2}\theta$

$$\theta = \left[180 - 2Sin^{-1}\left(\frac{D_2 - D_1}{2c}\right)\right]\frac{\pi}{180} = 2.71rad \quad \dots \quad (28)$$

Recall, D_1= 75 mm, D_2= 375 mm, c = 700 mm

Solving equations (26), (27), and (28) simultaneously yielded T_1= 204.04 N and T_2= 0.00342 N,

From Figure 2, the maximum bending moment was computed as:

M_b = 97.6046 × 325 = 31722.08 Nmm

Therefore, the size of shelling shaft was determined according to Khurmi and Gupta (2005).

$$d = \sqrt[3]{\frac{16}{\pi\tau_{allowable}}\sqrt{(K_bM_b)^2 + (K_tM_t)^2}} \quad \dots \quad (29)$$

$$d = \sqrt[3]{\frac{16}{\pi \times 56.25}\sqrt{(1.5 \times 31722.08)^2 + (1.5 \times 130.517)^2}}$$

$= 16.17 \approx 20$ mm

Where, d = diameter of the shelling shaft, $\tau_{allowable}$ = allowable shear stress for the shaft (56.25 N/mm^2), K_b = combined shock and fatigue factor as applied to bending moment (1.5 to 2.0) for suddenly applied load with minor shock, K_t = combined shock and fatigue factor as applied to torsional moment (1.0 to 1.5) for suddenly applied load, M_t = torsional moment (130.517 Nmm), M_b = bending moment (31722.08 Nmm).

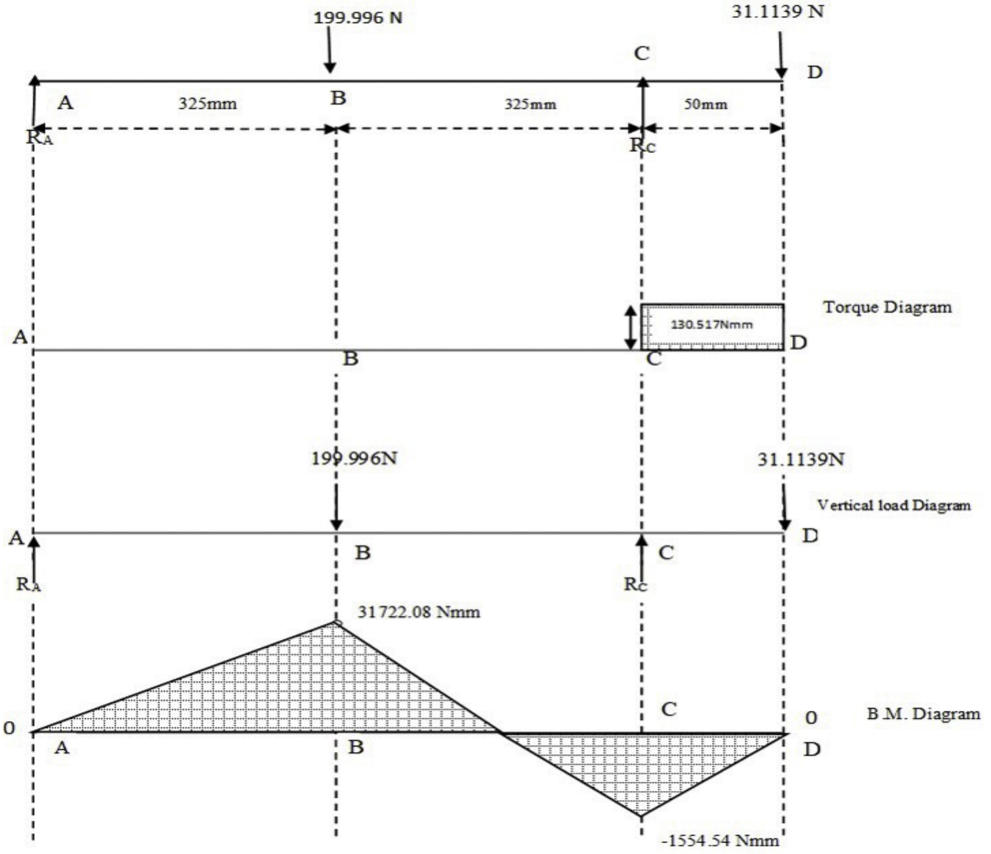

Figure 2: Space diagram, torque, loads and bending moment acting on the shelling shaft

2.5.2 Blower shaft

A similar procedure was repeated for determining the size of the blower shaft. To calculate the tension on the belt, the torsional moment was computed accordingly:

$$M_t = 9.55\frac{H}{N_3} = 9.55\frac{4.1 \times 10^3}{680} = 57.5809 \text{ Nmm}$$

$$(T_1 - T_2) \times r = 57.5809 \quad \ldots \quad (30)$$

Where, r = radius of driven pulley = 160 ÷ 2 = 80 mm.

The angle of contact made by the belt on the blower pulley was:

$$2.3log\left(\frac{T_1}{T_2}\right) = \mu \times \theta \times Cosec\beta \ \ldots \quad (31)$$

$$\theta = \left[180 - 2Sin^{-1}\left(\frac{D_3 - D_1}{2c}\right)\right]\frac{\pi}{180} = 2.86\, rad \quad \ldots \quad (32)$$

Recall, D_1= 75 mm, D_3= 160 mm, c = 300 mm, $\beta = \frac{1}{2}\theta$

Solving equations 30, 31, and 31 simultaneously yielded T_1= 0.7206 N and T_2= 0.00082 N

The bending moment was computed at point B (Figure 3),

$$M_b = 7.4821 \times 175 = 1309.38 \text{ Nmm}$$

Therefore, the size of the blower shaft is calculated as:

$$d = \sqrt[3]{\frac{16}{\pi \times 56.25}\sqrt{(1.5 \times 1306.38)^2 + (1.5 \times 5809)^2}}$$

$$= 8.662 \text{ mm} \approx 10 \text{ mm}$$

The standard size of 16 mm was selected

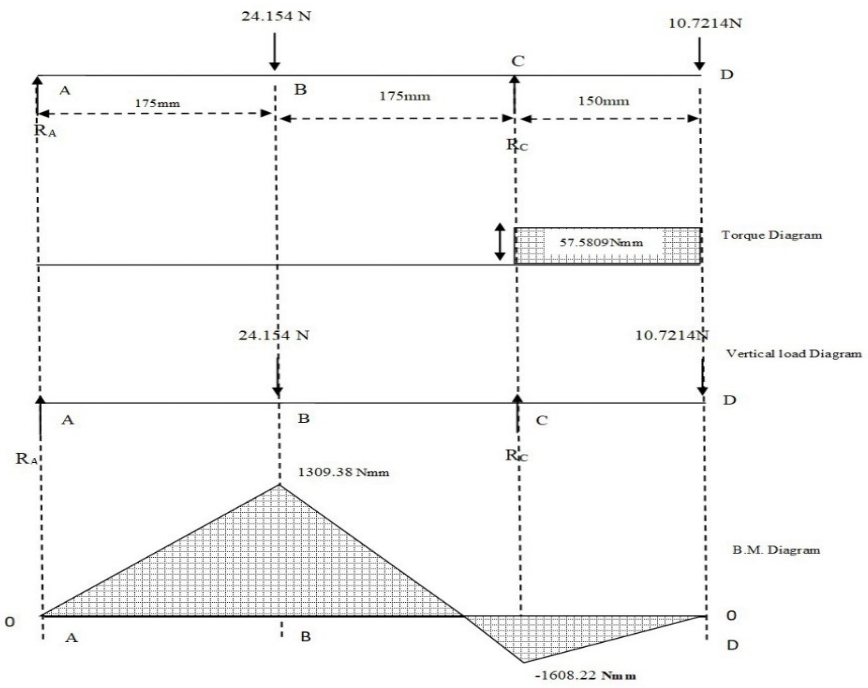

Figure 3: Space diagram, Torque, Loads and bending moment acting on the blower shaft

2.6 Selection of bearings
Standard bearings were selected based on their load-carrying capacity, life expectancy, and reliability as reported in bearing catalogue (SKF, 2012). Minimum shaft diameters were considered for both the shelling and blower shafts.

2.6.1 Selection of bearing for shelling shaft
The relationships between the basic rating life, the basic dynamic load rating (C), and bearing load (P) were determined according to Karwa (2010) as:

$$L_{10} = \frac{60\,n}{10^6} \times L_H \qquad \ldots \qquad (33)$$

$$C = (L_{10})^{1/k} \times P \qquad \ldots \qquad (34)$$

Where, L_{10} = rating life of bearing for 90% survival at one million revolutions, L_H = required life of bearing in million revolution, k = exponential for life equation of bearing (k = 3 for ball bearing), n = bearing rotational speed (rpm); $n = N_2$ = 300 rpm, L_H= 8000hrs, (SKF, 2012).
Since the shaft has a constant diameter, the highest radial load was used as equivalent dynamic bearing load (i.e. $P = R_C$ = 1001.07 N) for the bearing design.

$$L_{10} = \frac{60 \times 300}{10^6} \times 8000 = 144$$

$$C = (144)^{1/3} \times 1001.07 = 5247.01 \approx 5.2 \text{ kN}$$

According to the SKF bearing catalogue, the dynamic capacity of 6204 bearing was 13.5 kN, and this was greater than the required capacity. Hence such bearing was selected (SKF, 2012).

2.6.2 Selection of bearing for blower shaft
Since the shaft was of constant size, the highest radial load was used as equivalent dynamic bearing load ($P = R_C$ = 915.427 N) for the bearing design, L_H= 8000 hrs, k = 3 (for ball bearing), $n = N_3$ = 680 rpm, L_{10} = rating life of bearing for 90% survival at one million revolutions.

$$\therefore L_{10} = \frac{60 \times 680}{10^6} \times 8000 = 326.4, \text{ and}$$

$$C = (326.4)^{1/3} \times 915.43 = 6302.9 \text{ N} \approx 6.3 \text{ kN}$$

From the SKF bearing catalogue, the dynamic capacity of 6203 bearing was 9.95 kN, this was greater than the required capacity. As such, this bearing was selected (SKF, 2012).

2.7 Performance Test

Figure 4 showcased the old and new groundnut shellers after construction. For the performance test, *Ex-Dakar* groundnut variety was used. The groundnut was purchased from Dawanau market, in Kano State. Other input parameters used are drum speed of 180 rpm, moisture content of 8%, and a feed rate of 300 kg/h. The performance of the prototype modified groundnut sheller was evaluated based on output capacity, shelling efficiency, cleaning efficiency, mechanical damage, and scatter loss. Student t-test was used to compare the results.

Figure 4: Locally made groundnut sheller (A) and new groundnut sheller after modification (B)

2.7.1. Output Capacity

$T_p = \frac{W_c}{T_m}$ (Maduako, Saidu, Matthias, and Vanke, 2006) … (35)

Where, T_p= output capacity (kg/h), W_c= weight of shelled groundnut received at the main outlet (kg), T_m = time of shelling operation (hr).

2.7.2 Shelling efficiency

$\varepsilon_s = 1 - \frac{Q_u}{Q_T} \times 100$ (Alonge and Kosemani, 2011) … (36)

Where, ε_s= shelling efficiency (%), Q_T= total weight of groundnut sample (kg), Q_u = weight of unshelled groundnut (kg).

2.7.3 Cleaning efficiency

$C_\varepsilon = \frac{Q_s}{Q_s + Q_c} \times 100$ (Mohammed and Hassan, 2012) … (37)

Where, C_ε = cleaning efficiency (%), Q_c = weight of chaff collected at the main outlet (kg), Q_s = weight of cleaned groundnut kernels received at the outlet (kg).

2.7.4 Mechanical damage

$M_d = \frac{Q_D}{Q_g + Q_D} \times 100$ (Maduako et al., 2006) … (38)

Where, M_d = mechanical damage (%), Q_g = weight of undamaged groundnut seeds (kg), Q_D= weight of damage groundnut seed (kg)

2.7.5 Scatter loss

$W_G = \frac{Q_x}{Q_T} \times 100$ (Alonge and Kosemani, 2011) … (39)

Where, W_G= scatter loss (%), Q_x= weight of wasted groundnut kernels collected at chaff outlet (kg).

ISSN: 2449 – 0539
BAYERO JOURNAL OF ENGINEERING AND TECHNOLOGY (BJET) VOL.14 NO.2, pp 169-182, AUGUST, 2019

3.0 RESULTS AND DISCUSSION

The results of the performance test of the modified and old groundnut shellers are presented in Table 1 and Figure 5, respectively. It can be observed from the results that the modified sheller has an output capacity of 239.81 kg/h, which was higher than that of the old sheller (233.18 kg/h). Also, no significant difference in the output capacity was noticed among the two groundnut shellers. Output capacity of 345.4 kg/h was reported in an electrically operated groundnut sheller (Mohammed and Hassan, 2012).

Likewise, the shelling efficiency (98.32%) of the modified sheller was significantly higher than that recorded for the old sheller (86.19%). However, the old groundnut sheller had no cleaning unit. Therefore, the integrated cleaning unit in the modified sheller has an efficiency of 50.63%. Shelling efficiencies of 96.3% and 94.5% for locally developed and imported Kirlosker groundnut shellers were reported (Oduma, Edeh, & Eze, 2015).

One of the major problems of groundnut shelling is the kernel damage. In order to minimize kernel damage, proper design considerations such as the physical properties of the groundnut need to be considered. In our result, the mechanical damage of the old groundnut sheller was significantly reduced after modification (Figure 5). Mechanical damage of 4.33 % for the modified sheller was recorded as against 8.11% for the old sheller (Table 1). The decrease in mechanical damage could be attributed to the modification carried out on the shelling drum. That is, the angular iron used in the old sheller was replaced with cylindrical iron rods. Also, the precise determination of concave clearance based on groundnut properties (*Ex-Dakar* variety) during the design of the modified sheller could contribute to the low kernel damage (Muhammad, Ahmad, & Lawan, 2017). An improvement from 55.3% - 87% in the shelling efficiency of a wooden beater groundnut sheller was reported after modification (Gitau et al., 2013).

The scatter loss in the modified sheller was significantly lower than that obtained in the old sheller (Figure 5). The scatter loss for the modified sheller was 3.24% as compared with 9.52% for the old sheller. In the modified machine, majors were carefully taken during the construction to adhere to the design concept in order to reduce kernels scattering. Among which was providing a flood gate control which served as feed control as well as preventing groundnut from spilling out from the shelling chamber. Following the modification of a groundnut sheller by Helmy and co-researchers, an increase in shelling efficiency from 95.32 – 98.85%, reduction in seed damage from 6.12 – 1.36%, and a decrease in total losses from 10.8 – 2.51% were obtained (Helmy et al., 2013).

Table 1: Performance parameters of the modified and old groundnut shellers.

Modified groundnut sheller					
Replication	**Output capacity (kg/h)**	**Shelling efficiency (%)**	**Cleaning efficiency (%)**	**Mechanical damage (%)**	**Scatter loss (%)**
1	238.76	98.22	51.91	4.35	3.54
2	239.86	98.42	49.87	5.15	3.43
3	240.81	98.32	50.12	3.48	2.75
Mean	239.81	98.32	50.63	4.33	3.24
Old groundnut sheller					
1	228.11	88.26	-	7.55	9.56
2	236.79	80.08	-	8.00	8.99
3	234.65	90.23	-	8.79	10.01
Mean	233.18	86.19	-	8.11	9.52

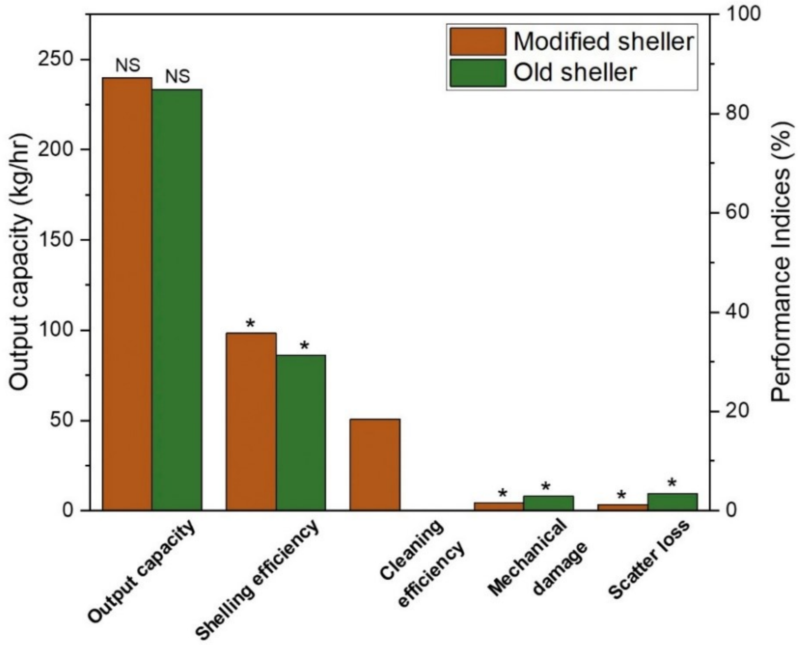

Figure 5: Comparative performance indices of the modified and old groundnut shellers using the t-test. The * indicate significant difference at 5 % and NS refers to non-significant difference.

4.0 CONCLUSION

A local groundnut sheller was modified to improved performance and incorporate cleaning unit. The sheller was constructed from locally available materials with a compact and robust design that suit the local requirements. The performance test revealed that the modified sheller was superior to the old sheller using *Ex-Dakar* groundnut variety at a moisture content of 8%, 300 kg/h feed rate, and 180 rpm drum speed. The modified sheller has a mean output capacity of 239.81 kg/h, shelling efficiency of 98.32 %, cleaning efficiency of 50.63%, mechanical damage of 4.33%, and scatter loss of 3.24% under the input parameters mentioned above. Nevertheless, the performance indices can be further improved through optimization of the input parameters. It is therefore recommended to carry out an optimization of performance of the groundnut sheller in order to obtain optimum operating conditions.

ACKNOWLEDGEMENT

The authors acknowledged the financial support for the graduate study from Bayero University, Kano, Nigeria.

REFERENCES

Alonge, A. F., & Kosemani, B. S. (2011). Development of a guinea corn thresher. In *A paper written for presentation at the 2011 ASABE Annual International Meeting sponsored by ASABE Galt House Hotel Louisville, Kentucky. Paper Number: 1111268.* (pp. 1–8). USA: ASABE. https://doi.org/DOI: 10.13031/2013.38497

Asiedu, J. J. (1992). *Processing of Tropical Crops - A Technological Approach*. London, Basingstoke: Macmillan Press limited.

Butts, C. L., Sorensen, R. B., Nuti, R. C., Lamb, M. C., & Faircloth, W. H. (2009). Performance of equipment for in-field shelling of peanut for Biodiesel production. *Transactions of the American Society of Agricultural and Biological Engineers*, *52*(5), 1461–1469.

Ghanem, T. H., & Shetawy, M. E. (2009). Some mechanical properties of hulled peanuts and kernels. In *Agricultural engineering and variables of the present epoch* (pp. 1586–1597).

Gitau, A. N., Mboye, P., Njoroge, B. N. K., & Mburu, M. (2013). Optimizing the performance of a manually operated groundnut (Arachis hypogaea) decorticator. *Open Journal of Optimization*, *02*, 26–32. https://doi.org/10.4236/ojop.2013.21004

Gui, Y., & Xi, P. (2010). Study on the methods of determining main geometric parameters of centrifugal fan impeller. *IEEE*, *4*, 42–46.

Helmy, M. A., Abdallah, S. E., Mitrroi, A., & Basiouny, M. A. (2013). Modification and performance evaluation of a reciprocating machine for shelling peanut. *Misr Journal of Agricultural Engineering*, *24*(2), 283–298.

IAR. (2005). Operation and maintenance of IAR groundnut shellers. *Samaru Extension Bulletin. Ahmadu Bello University, Zaria. Mechanization Series No. 1.*

Ibrahim, A. and Onwualu, A.P., (2005). Technologies for Extraction of Oil from Oil-Bearing Agricultural Products: A Review. *Journal of Agricultural Engineering and Technology (JAET)*. Vol.13. Pp 58 – 89.

Joshi, H. C. (1981). Design and selection of thresher parameters and components. *Agricultural Mechanization in Asia, Africa and Latin America*, *2*(12), 61–68.

Karwa, R. (2010). *A textbook of machine design* (2nd ed.). New Delhi: Laxmi Publications (p) Ltd.

Khurmi, R. S., & Gupta, J. K. (2005). *A textbook of machine design* (first). New Delhi, India: Eurasia publishing company (PVT) Ltd.

Maduako, J. N., Saidu, M., Matthias, P., & Vanke, I. (2006). Testing of an engine-powered groundnut shelling machine. *Journal of Agricultural Engineering and Technology*, *14*, 29–37.

Mohammed, A., & Hassan, A. B. (2012). Design and evaluation of a motorized and manually operated groundnut shelling machine. *International Journal of Emerging Trends in Engineering and Development*, *4*(2), 673–682.

Mohtasebi, S. S., Lar, B. M., Alidadi, J., & Besharati, K. (2006). A new design for grain combine thresher. *International Journal of Agriculture and Biology*, *8*(5), 680–683.

Muhammad, A. I., Ahmad, R. K., & Lawan, I. (2017). Effect of moisture content on some engineering properties of groundnut pods and kernels. *Agricultural Engineering International: CIGR Journal*, *19*(4), 200–208.

Muhammad, A. I., Attanda, M. L., Mustapha, A., & Inuwa, M. B. (2015a). Waste management and utilization in Dawakin Tofa Local Government of Kano State. In *Proceeding of the 49th Annual Conference of the Agricultural Society of Nigeria*. Delta State University, Delta: Agricultural Society of Nigeria.

Muhammad, A. I., Isiaka, M., Fagge, A. A., Attanda, M. L., Lawan, I., & Dangora, N. D. (2015b). Some engineering properties of three varieties of groundnut pods and kernels. *Arid Zone Journal of Engineering, Technology and Environment.*, *11*, 61–75.

Muhammad, J. (2009). *Development of maize dehusker thresher*. Ahmadu Bello University, Zaria, Nigeria.

Nalado, D. D. (2006). *Modification of IAR maize dehusker sheller for multi crop threshing*. Ahmadu Bello University, Zaria, Nigeria.

Ntare, B., Diallo, A., Ndjeunga, J., & Waliyar, F. (2008). *Groundnut seed production manual*. https://doi.org/10.1017/CBO9781107415324.004

Odesanya, K. O., Adebiyi, K. A., & Salau, T. A. O. (2015). Estimation of engineering parameters for the development of a groundnut decorticator. *International Journal of Novel Research in Engineering and Applied Sciences*, *2*(1), 2–25.

Oduma, O., Edeh, J. C., & Eze, C. C. (2015). Comperative performance of a locally developed groundnut decorticator with an imported Kirlosker decorticator. *Internation Journal of Engineering and Technology*, *5*(4), 205–211.

Oyelami, A. T., Olaniyan, O. O., Iliya, D., & Idowu, A. S. (2008). The design of a closed-type-impeller blower for a 500kg capacity rotary furnace. *AU Journal of Technology*, *12*(1), 50–56.

Ramli, R. H. M. (2003). *Designing and building a peanut-shelling machine to help people in third world countries such as Ghana/Haiti*. Massachusetts Institute of Technology.

Sahay, K. M., & Singh, K. K. (2007). *Unit operations of agricultural processing* (2nd ed.). New Delhi: UBS publishers Distributors, PVT LTD.

Shigley, J. E., & Mischke, C. R. (2001). *Mechanical engineering design* (6th ed.). Michigan: McGraw Hill.

SKF. (2012). *Rolling Bearings Catalogue*. 4955 Gulf Freeway, Houston, TX 77023: SKF Group.

Ugwuoke, I. C., Okegbile, O. J., & Ikechukwu, I. B. (2014). Design and fabrication of groundnut shelling and separating machine. *International Journal of Engineering Science Invention*, *3*(4), 60–66. Retrieved from www.ijesi.org